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The partially-coupled modal contribution assumption of noise radiation and the dominant noise-contribution mode

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ABSTRACT

A novel partially-coupled modal contribution assumption is presented in this paper. The partially-coupled assumption is theoretically presented, and it is proved to be more reasonable according to the law of conservation of energy compared with the traditional decoupled and coupled assumptions. The proposed method is applied to analyze a complex engineering structure. Based on the partially-coupled assumption, the dominant noise-contribution mode (DNCM) is identified at each frequency. The DNCM method is more effective in determining the most significant mode which makes the noise control more precise.

1. Introduction

With the wide application of computer aided engineering (CAE) technology, it is now possible for the vibro-acoustic predictions and NVH performance optimizations of structural components before manufacturing [1–3]. The modal superposition method (MSM) rather than the direct numerical integration (DNI) method is preferred to calculate the forced vibration responses because of its high efficiency [4–6]. The finite element method (FEM) and boundary element method (BEM) are usually combined to calculate the vibro-acoustic characteristics in low frequency based on the vibration responses [7]. The spectral analysis is usually applied to the interpretation of the simulated results to determine the specific optimization targets. The conventional procedure is to find out the modes located close to the frequencies of the dominant noise peaks and suppress their modal shapes by using structural optimization methods, such as the topology optimization and the topography optimization [8–11].

In theory, the relation between the modes and vibration responses is linear according to MSM, while the derivational process from vibration responses to the radiation noise is nonlinear, which indicates that the coupling relation among the modes in the noise generation is very complicated. Since the modal frequency is not directly corresponding to the noise peak, it is impractical to determine the critical modes simply by the frequency-domain characteristics of the acoustic results. What is more, this approach would decrease the work efficiency, or even lead to a wrong solution with opposite effects in some cases.

Natarajan [12] proposed the forced vibro-acoustic components (F-VAC) method using the decoupled modal contribution assumption and the coupled modal contribution assumption, where the acoustic contributions of the modes are assumed to

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be decoupled and coupled. The F-VAC approach showed the complexity and nonlinearity of the relation between the vibration responses and the radiation noise, and it indicated that it was more cost-effective using the coupled modal contribution assumption in noise control.

In this paper, a novel partially-coupled modal contribution assumption is proposed. Firstly, the two assumptions proposed by Natarajan will be analyzed and a more reasonable assumption called partially-coupled modal contribution assumption which perfectly achieves the conservation of energy will be presented. Secondly, a numerical analysis will be applied to a complex structure commonly used in engineering to validate the proposed method. The numerical results will be compared with the results using commercial finite element analysis (FEA) software. The comparisons indicate the proposed method is of enough accuracy. Then the advantage of the partially-coupled modal contribution assumption will be proved, and the dominant noise-contribution mode (DNCM) at every frequency will be calculated based on the partially-coupled assumption. DNCM could be used to explain the nonlinear relation between the modes and the radiation noise intuitively, and enhance the precision and efficiency of the structural optimization for the low noise performance.

2. Vibration and acoustic theories

2.1. Forced vibration responses

The mathematical purpose of the response analysis is to solve the second order system of ordinary differential equations

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F}(t) \tag{1}$$

where, **M** and **K** are the structural mass matrix and stiffness matrix; **C** is the Rayleigh damping matrix, which is written as $\mathbf{C} = \alpha\mathbf{M} + \beta\mathbf{K}$ for the convenience of decoupling; **x** is the structural displacement vector; $\mathbf{F}(t)$ represents the exciting force matrix in the time domain.

MSM is applied to solve the above equation where **x** can be regarded as a superposition of the normal modes $\mathbf{\Phi}$ according to the expansion theorem, written as

$$\mathbf{x} = \mathbf{\Phi}\mathbf{q} \tag{2}$$

where, $\mathbf{q} = \begin{bmatrix} q_1 & q_2 & \cdots & q_n \end{bmatrix}^{\mathrm{T}}$ is the vector consisting of modal participation factors (MPFs), n represents the number of modes. Introducing Eq. (2) to Eq. (1), and pre-multiplying both sides of the equation by $\mathbf{\Phi}^{\mathrm{T}}$, it becomes

$$\mathbf{\Phi}^{\mathrm{T}}\mathbf{M}\mathbf{\Phi}\ddot{\mathbf{q}} + \mathbf{\Phi}^{\mathrm{T}}\mathbf{C}\mathbf{\Phi}\dot{\mathbf{q}} + \mathbf{\Phi}^{\mathrm{T}}\mathbf{K}\mathbf{\Phi}\mathbf{q} = \mathbf{\Phi}^{\mathrm{T}}\mathbf{F}(t) \tag{3}$$

According to the orthogonality of modal vectors [13], $\mathbf{\Phi}^{\mathrm{T}}\mathbf{M}\mathbf{\Phi} = diag\begin{pmatrix} M_1 & M_2 & \cdots & M_r & \cdots & M_n \end{pmatrix}$, $\mathbf{\Phi}^{\mathrm{T}}\mathbf{C}\mathbf{\Phi} = diag\begin{pmatrix} C_1 & C_2 & \cdots & C_r & \cdots & C_n \end{pmatrix}$, $\mathbf{\Phi}^{\mathrm{T}}\mathbf{K}\mathbf{\Phi} = diag\begin{pmatrix} K_1 & K_2 & \cdots & K_r & \cdots & K_n \end{pmatrix}$ and let $\mathbf{\Phi}^{\mathrm{T}}\mathbf{F}(t) = \begin{bmatrix} F_1 & F_2 & \cdots & F_r & \cdots & F_n \end{bmatrix}^{\mathrm{T}}$, where $M_r = \mathbf{\Phi}_r^{\mathrm{T}}\mathbf{M}\mathbf{\Phi}_r$, $C_r = \mathbf{\Phi}_r^{\mathrm{T}}\mathbf{C}\mathbf{\Phi}_r$, $K_r = \mathbf{\Phi}_r^{\mathrm{T}}\mathbf{K}\mathbf{\Phi}_r$, $F_r = \mathbf{\Phi}_r^{\mathrm{T}}\mathbf{F}(t)$, which are called modal mass, modal damping, modal stiffness and modal force of the rth mode, respectively. In this way, for the rth mode, Eq. (3) becomes

$$M_r\ddot{q}_r + C_r\dot{q}_r + K_r q_r = F_r \tag{4}$$

Thus, **q** can be easily solved, and **x** can be obtained using Eq. (2).

2.2. Radiation noise

For a higher computational efficiency, the linear acoustic elements close to the radiation surface are established with the length larger than the structural elements but less than 1/6 of the wavelength corresponding to the concerned maximum frequency. The structural vibration information of the surface nodes are mapped to these acoustic elements with the application of weighted average algorithm. The calculation principle of the velocity mapping is: (1) Set a modest mapping distance D and find out all the structural nodes in the mapping space A which is a sphere of radius D centered at the acoustic

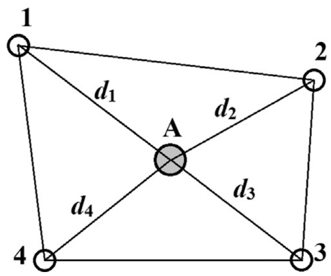

Fig. 1. Mapping schematic of the vibration information.